NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL NOTE 2426

AN INVESTIGATION OF AIRCRAFT HEATERS

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SUMMARY

An experimental investigation has been performed on the thermal and hydrodynamical behavior of air flowing along a passage of which one side was flat (and steam-heated) and the other side was wave-shaped (and unheated).

Two series of tests were conducted: One series in which the wave-shaped plate and the steam-heated plate were placed in exact opposition (thermal boundary layer at leading edge of flat plate and flow over wave plate beginning at same point) and the other series in which the leading edge of the wave plate was placed 9 inches upstream of the leading edge of the heated plate, thereby promoting turbulence upstream of the beginning of the thermal boundary layer. In the first series of tests, two wave plates of different amplitude and wavelength were used. A range of air weight rates and spacings between the heated plate and the wave plate was investigated.

An analysis of the heat-transfer and pressure-drop data reveals that the turbulence produced by air flowing over wavy surfaces approximately doubles the heat-transfer rate for narrow duct widths. Of course, at large duct widths the effect is negligible. The pressure drop was greatly increased over that for a straight duct. The pressure drop was increased to at least 4 times for large duct widths and to at least 10 times for small duct widths.

INTRODUCTION

Extensive tests on aircraft heat exchangers carried out in this laboratory under the auspices of the National Advisory Committee for Aeronautics indicated that some systems were easily predictable from established empirical heat-transfer equations. However, many systems encountered were not readily predictable. These systems consisted of some finned surfaces and certain cases in which the passage was not straight and of uniform cross-sectional area. A finned-surface system was investigated and reported in reference 1. A study of heat-transfer and pressure-drop data for the case of air flow between a heated plane surface and an unheated wavy surface is presented herein.

One object of the investigation was to determine the added heat-transfer rate and pressure drop caused by the sinuous flow. The experimental data were thus compared with the results for flow between two plane surfaces. Of course, the increase of the heat-transfer rate due to the presence of the wave plate was found to vary with the air weight rate, the passage width, the amplitude and wave-length of the wavy surface, and the distance from the origin of the thermal boundary layer (beginning of heated section).

The unit thermal conductance at the leading edge of a heated flat plate in a fluid stream of uniform velocity is theoretically infinite and decreases with increasing distance from the leading edge. For this case, the hydrodynamic boundary layer and the thermal boundary layer begin at the same point. Experiments reported in references 1 and 2 for flow inside ducts showed that high values of the unit thermal conductance were obtained near the origin of the thermal boundary layer even for the case in which the hydrodynamic boundary layer was initiated at a point several diameters upstream.

An investigation of this phenomenon was carried on for the present system by making two series of tests: One series in which the thermal boundary layer and the turbulence-generating wave plate were initiated at the same point by placing the steam-heated plate in exact opposition to the wavy plate (fig. 1, systems (1) and (2)) and the other series in which the turbulence was initiated before the thermal boundary layer by placing the leading edge of the wavy plate 9 inches ahead of the steam-heated plate (fig. 1, system (3)).

This investigation, part of a research program conducted at the University of California, was sponsored and financed by the NACA.

SYMBOLS

A	heat-transfer area corresponding to measured heat rate, square feet
D_H	hydraulic diameter, feet $\left(\frac{4 \times \text{Cross-sectional area}}{\text{Wetted perimeter}}\right)$
$f_{c_{av}}$	average unit thermal conductance, Btu/(hr)(sq ft)(°F)
f_{c_x}	local or point value of unit thermal conductance, Btu/(hr)(sq ft)(°F)
g	gravitational force per unit mass, 32.2 (lb)/(lb-sec^2)/(ft)
G	average weight rate of fluid per unit cross-sectional area, (lb)/(hr)(sq ft)
Δh_{vap}	heat of vaporization for water at atmospheric pressure and 212° F, Btu/(lb)
K	static-pressure-drop coefficient (see equation (4))
Δp	static-pressure drop between two points of duct, (lb)/(sq ft)
q	heat-transfer rate, Btu/(hr)
R_{av}	average rate of condensation per condensing steam section under "load" conditions, (lb)/(hr)
R'_{av}	average rate of condensation per condensing steam section under "no-load" conditions, (lb)/(hr)
t_p	measured surface temperature, °F
T	mixed-mean air absolute temperature, °R
T_f	arithmetic average of mixed-mean absolute temperature of air and absolute temperature of plate, °R
u_{max}	velocity of air taken at minimum cross-sectional area in duct, (ft)/(sec)

x distance along heated plate, feet

y_{min} minimum duct width; distance from heated plate to crest of wave plate, inches

y_o duct width; distance from heated plate to hollow of wave plate directly opposite, inches

γ air density, (lb)/(cu ft)

τ_o mixed-mean temperature of air at entrance of test section, $^{\circ}F$

τ_x computed local mixed-mean air temperature, $^{\circ}F$

DESCRIPTION OF APPARATUS

The test equipment was the same steam-condensing calorimetric apparatus as that used in the pin-fin tests described in reference 1. The steam-heated plate used here was the so-called "unfinned" plate used in those tests. The back of this plate was fitted with eight steam-condensate collection sections in order to obtain local or point heat-transfer rates. The wave plate was mounted on the opposite side of the duct on the panel which was movable in order to vary the distance between the two plates (see fig. 1 for schematic drawing of equipment).

The dimensions of the wave plates and their positions for the various systems may be given as follows:

	System I	System 2	System 3
Wave plate used	Small	Large	Large
Double amplitude, in.	0.22	0.62	0.62
Wave length, in.	1.3	2.6	2.6
Total length of wave plate, in.	20	31	31
Position of leading edge of wave plate with respect to leading edge of heated plate	Directly opposite	Directly opposite	9 in. upstream

Both wave plates were made of corrugated sheet iron.

METHOD OF ANALYSIS

The procedure followed in obtaining the data for this report was the same as that described in reference 1.

Heat Transfer

From the data obtained, the local unit thermal conductance f_{c_x} is determined as a function of the distance from the leading edge of the heated test plate, the duct width (gap width) y_o, and the weight rate per unit area of air flowing through the passages G.

The heat transferred from the condensing steam in the steam chest to the air flowing through the passage is calculated from:

$$q = \Delta h_{vap}\left(R_{av} - R'_{av}\right) \tag{1}$$

where Δh_{vap} is the heat of vaporization for water at atmospheric pressure and 212° F, R_{av} is the average rate of condensation per section under "load" conditions, and R'_{av} is the average rate of condensation under "no-load" conditions. The terms "load" and "no load" apply to the test conditions in which air was passed and not passed, respectively, over the test plate while condensation data were obtained.

The mixed-mean temperatures of the air at the entrance τ_o and at the exit of the test section were measured. The temperature at the exit, together with the temperatures at intermediate points along the test section, was also calculated from the measurements of heat rates and air weight rates.

The unit thermal conductance is then calculated from

$$q = f_{c_x}A\left(t_p - \tau_x\right) \tag{2}$$

or

$$f_{c_x} = \frac{q}{A\left(t_p - \tau_x\right)} \tag{3}$$

where A is the heat-transfer area corresponding to the measured heat rate q, t_p is the measured surface temperature, and τ_x is the computed local mixed-mean air temperature.

The measured unit thermal conductances are compared with predicted values by means of the equation (reference 3) for

$$0 < \frac{x}{D_H} < 8.8 \quad \text{(see also reference 1)}$$

$$f_{c_x} = 7.3 \times 10^{-4}\, T_f^{0.3}\, \frac{G^{0.8}}{x^{0.2}}$$

for

$$\frac{x}{D_H} > 8.8$$

$$f_{c_x} = 5.4 \times 10^{-4}\, T^{0.3}\, \frac{G^{0.8}}{D_H^{0.2}}$$

where

x distance along heated plate, feet

D_H hydraulic diameter, feet $\left(\frac{4 \times \text{Cross-sectional area}}{\text{Wetted perimeter}}\right)$; $(D_H = 2y_o$ for small values of $y_o)$

f_{c_x} local or point value of unit thermal conductance, Btu/(hr)(sq ft)($^{\circ}$F)

T_f arithmetic average of mixed-mean absolute temperature of air and absolute temperature of plate, $^{\circ}$R

G average weight rate of fluid per unit cross-sectional area, (lb)/(hr)(sq ft)

T mixed-mean air absolute temperature, $^{\circ}$R

Static-Pressure Drop

The measured static-pressure drops have been corrected to apply to $16\frac{3}{4}$ inches of wave plate.

A static-pressure-drop coefficient was calculated from the equation:

$$\frac{\Delta p}{\gamma} = K\left(\frac{{u_{max}}^2}{2g}\right) \tag{4}$$

where u_{max} is the velocity of air taken at the minimum cross-sectional area for flow.

RESULTS AND DISCUSSION

Heat Transfer

The values of the local unit thermal conductance f_{c_x} are presented in figures 2 to 4 as a function of the distance along the heated plate x, the average weight rate per unit cross-sectional area G, and the duct width y_o. Two flow conditions were tested: One in which the heated length and the wave plate were in exact opposition (systems (1) and (2)) and one in which the flow over the wave plate was initiated upstream of the heated length (system (3)). In the former case two plates of different wavelength and amplitude were tried. (See fig. 1.) Included in each figure is a schematic diagram showing the flow condition for which the data were taken and the relative magnitudes of the duct width and of the amplitude and wavelength of the wave plate used.

The average unit thermal conductance $f_{c_{av}}$ is plotted in figure 5 against the average weight rate per unit cross-sectional area G, and figure 6 gives the variation of $f_{c_{av}}$ with the duct widths y_o for the three experimental conditions.

Upon reviewing the data obtained and the cross plots, the effect of the wave plate on the heat transfer over the heated plate may be summarized as follows: The increase of heat transfer along the heated plate (called the unfinned plate in reference 1) due to the presence of the wave plates is large for narrow duct widths only. This increase is due to the added turbulence and eddying flow and the alternate increase and decrease in fluid velocity caused by the wave plates. The effect of turbulence and eddying is evident in figure 7 where the variation of the local unit thermal conductance with distance along the heated plate is plotted at one weight rate per unit area and one narrow duct width for the three conditions:

(1) A heated plate without any wave plates (called unfinned flat plate in reference 1)

(2) The steam-heated plate and the wave plate placed in exact opposition (thermal boundary layer and flow over wave plate beginning at the same point, systems (1) and (2))

(3) The leading edge of the wave plate placed 9 inches upstream of the leading edge of the heated plate (system (3))

In the last case the amount of increased turbulence and eddy flow is practically constant along the heated plate and thus the increase of heat transfer in this case compared with the heated plate alone (case (1) above) is uniform along the test plate. The heat-transfer rate is approximately double that for the heated plate alone for the very narrow duct widths. At large duct widths the heat rate was about equal to that over flat plates alone, and in some cases lower than the flat-plate data presented in reference 1. The reason for this fact is not known. Since these data were obtained, however, some undeterminable errors in the measurement of the air weight rate due to small leaks in the apparatus were discovered. These errors may explain, in part, the discrepancy noted.

In the case where the steam-heated plate and the wave plate are initiated together, the heat-transfer rate is shown (fig. 7) to follow that of the unfinned plate for a short distance along the plate. After a transition period the heat-transfer rate increases to that of system (3) where the amount of added turbulence and eddying flow has approached an asymptote.

An examination of the distribution of heat-transfer rate near the entrance to the heated section reveals that the unit thermal conductance varies like that over a flat plate; that is, it approaches infinity at the leading edge, even though the hydrodynamic boundary layer has already started. This fact confirms the observations made in references 1 and 2.

Static-Pressure Drop

The static-pressure drop across the heated plate with different wave-plate conditions is presented in figure 8 as a function of the average weight rate per unit area G with the duct width y_o as parameter. The variation of the static-pressure drop with duct width is shown in figure 9.

The static-pressure drop increases with decreasing duct width at a greater rate than the heat-transfer rate (cf. corresponding curves in figs. 6 and 9).

An examination of figure 9 reveals that the pressure drop is greatly increased upon insertion of a wavy plate in a straight duct system. For small duct widths the increase was at least 10 times, while for large duct widths the increase was at least 4 times.

Attempts were made to predict the pressure drop along the wave-plate section by considering the gradual expansion and contraction of the air as it passed along each wavelength of the test section. The results obtained by this method were many times greater than the measured values.

The table below presents values of the pressure-drop coefficient K evaluated from equation (4).

System	y_o (in.)	y_{min} (in.)	K per wave
(1) - Short wave	5/8	1/2	0.07
(1) - Short wave	$3\frac{1}{8}$	$2\frac{7}{8}$	.01
(2) and (3) - Long wave	7/8	1/4	.12
(2) and (3) - Long wave	$3\frac{1}{8}$	$2\frac{1}{2}$	.04

Values of K estimated from expansion and contraction data were usually at least four times the values listed above. The term y_{min} is the minimum distance between the flat heated plate and the wave-shaped plate. The quantities y_o and y_{min} differ by the value of the double amplitude of the waves.

CONCLUSIONS

From an experimental investigation of the thermal and hydrodynamical behavior of air flowing between a flat and a wave-shaped plate, the following conclusions are drawn:

1. Turbulence caused by air flowing between a heated flat surface and a wavy unheated surface increases the heat transferred from the flat plate as much as twofold for narrow duct spacings over that for flow between two flat plates. This phenomenon does not occur, however, at wide duct widths.

2. The values of the local unit thermal conductance near the leading edge of the heated plate when the wave-shaped plate is placed upstream of the heated plate are higher than those for the case where the plates are placed in exact opposition because of the generated turbulence of the air.

3. The static-pressure drop for flow along the wave-shaped plate is 4 to 10 times (depending on the duct width and shape of plate) that for flow along straight ducts.

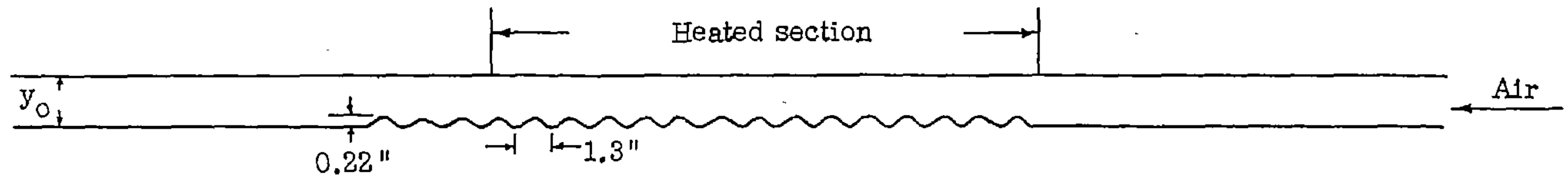
Department of Engineering
University of California
Berkeley, Calif., January 10, 1946

REFERENCES

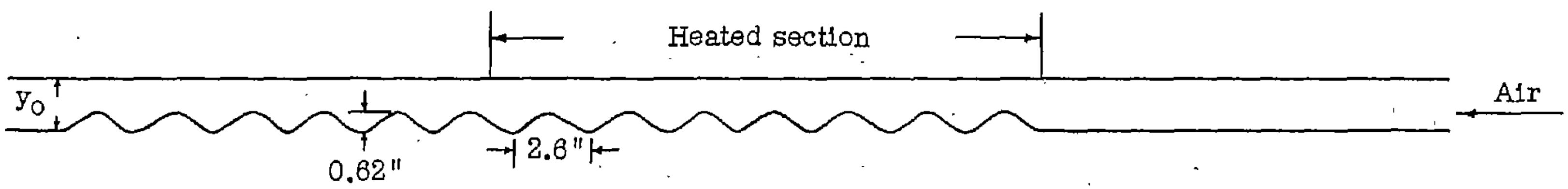
1. Boelter, L. M. K., Leasure, R., Romie, F. E., Sanders, V. D., Elswick, W. R., and Young, G.: An Investigation of Aircraft Heaters. XXXIII - Experimental Determination of Thermal and Hydrodynamical Behavior of Air Flowing along Finned Plates. NACA TN 2072, 1950.

2. Boelter, L. M. K., Young, G., and Iversen, H. W.: An Investigation of Aircraft Heaters. XXVII - Distribution of Heat-Transfer Rate in the Entrance Section of a Circular Tube. NACA TN 1451, 1948.

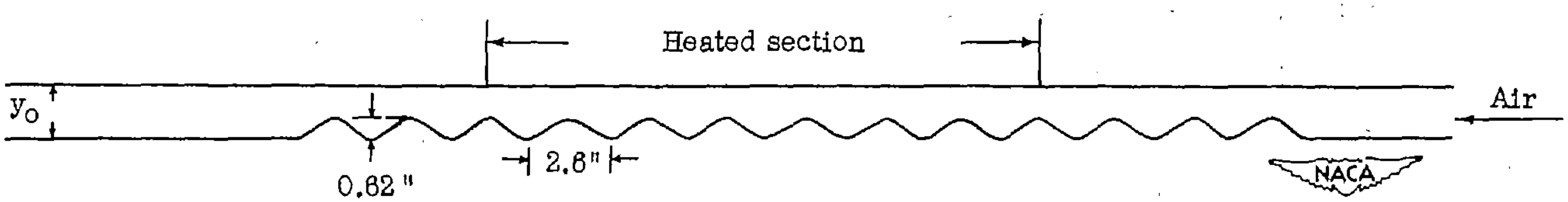
3. Boelter, L. M. K., Martinelli, R. C., Romie, F. E., and Morrin, E. H.: An Investigation of Aircraft Heaters. XVIII - A Design Manual for Exhaust Gas and Air Heat Exchangers. NACA ARR 5A06, 1945.



(a) System (1), small wave plate.

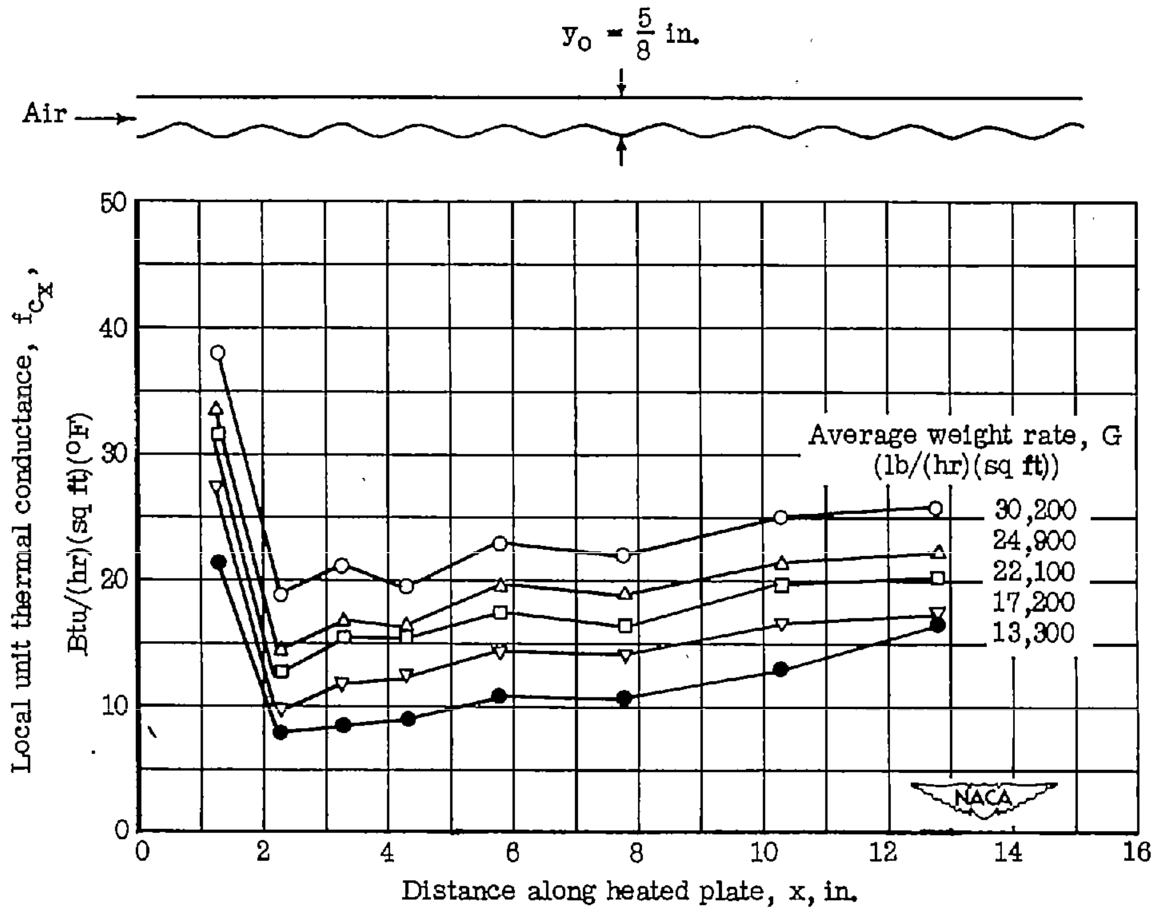


(b) System (2), large wave plate.



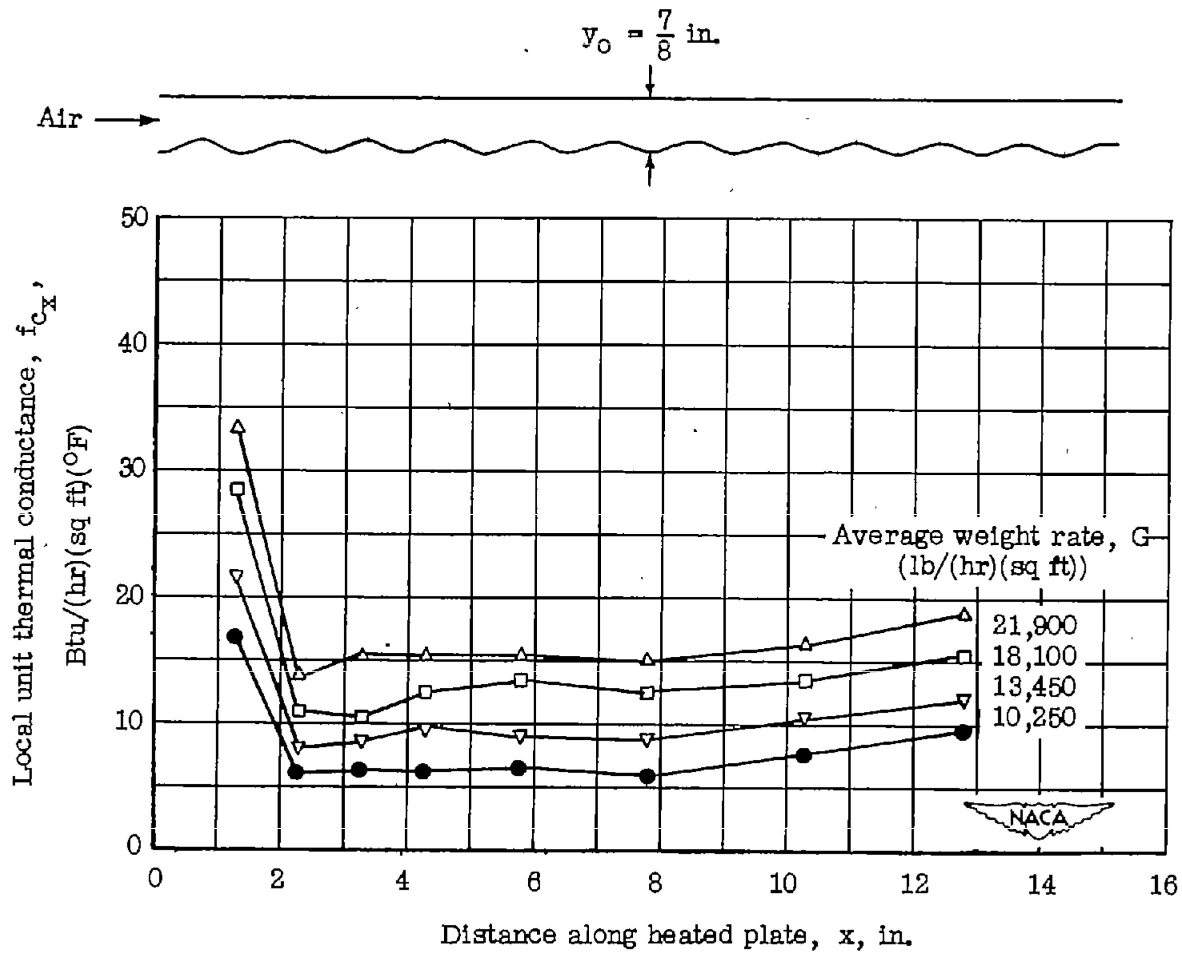
(c) System (3), large wave plate (leading edge upstream).

Figure 1.- Schematic diagram of three systems of aircraft heat exchangers.



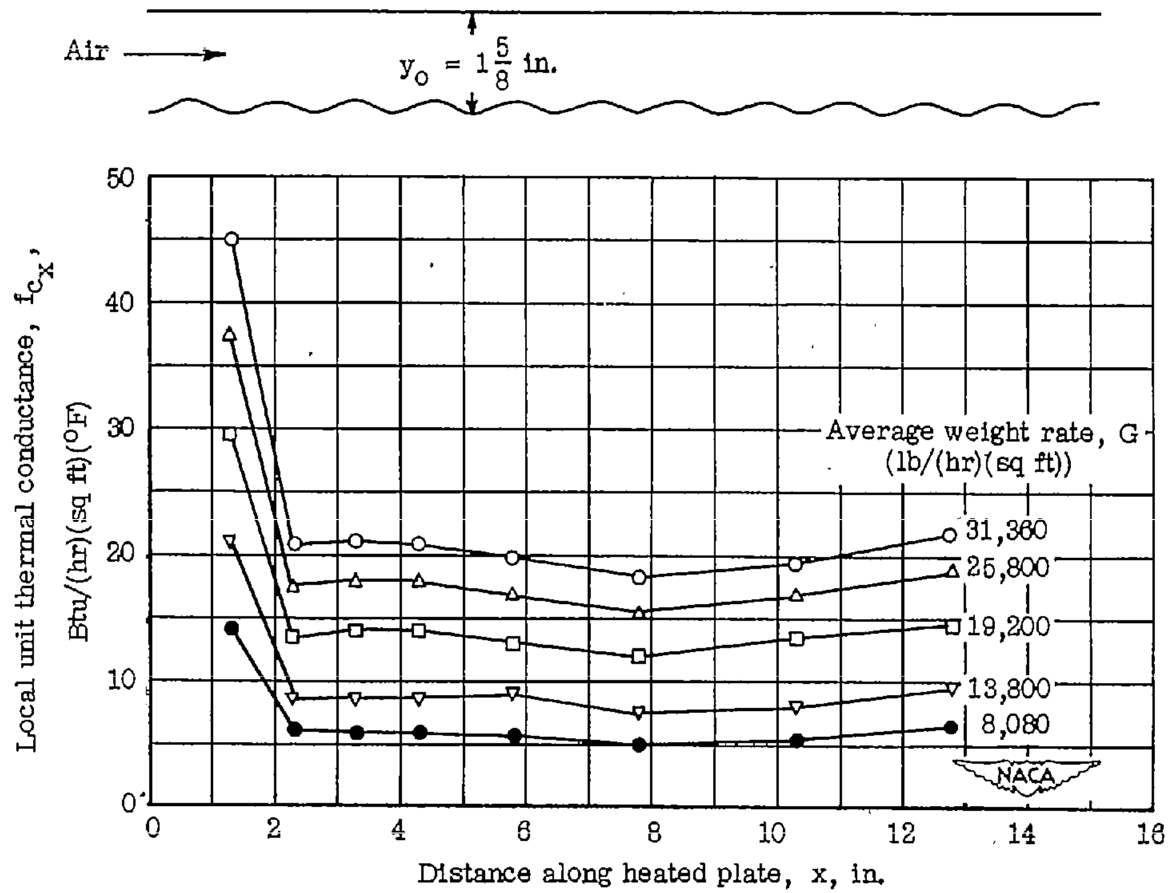
(a) Small wave plate at y_o = 5/8 inch.

Figure 2.- Variation of local unit thermal conductance with distance along heated plate for system (1).



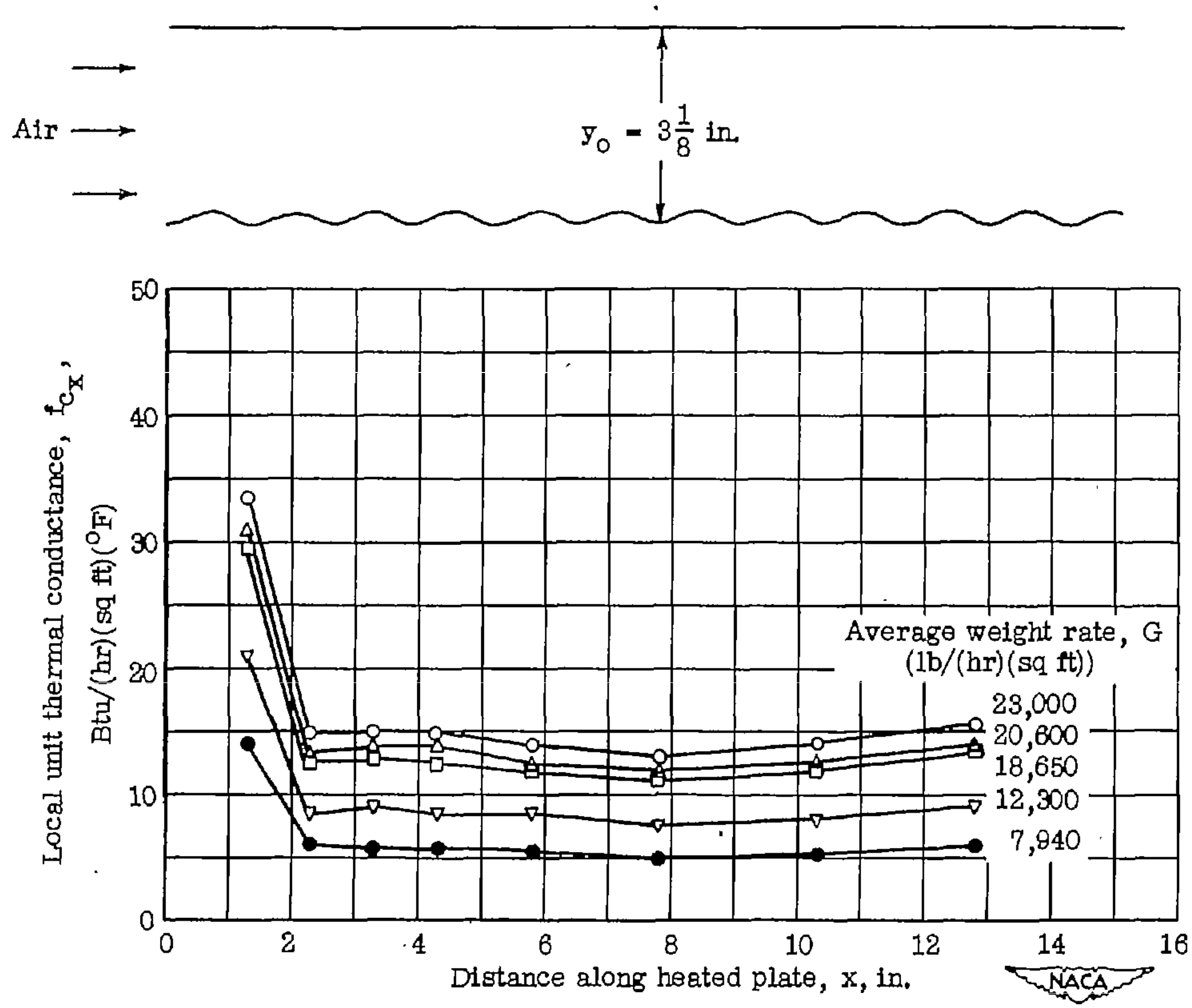
(b) Small wave plate at y_o = 7/8 inch.

Figure 2.- Continued.



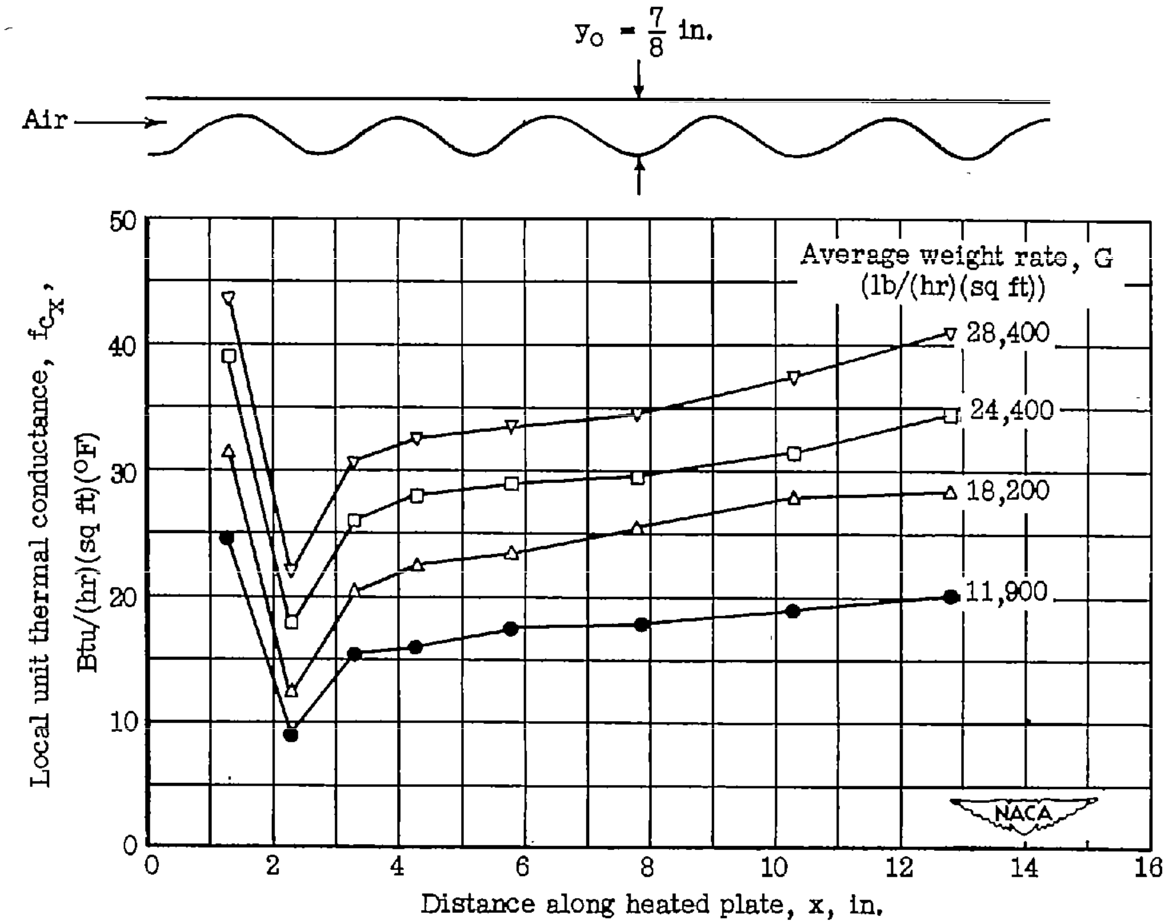
(c) Small wave plate at $y_o = 1\frac{5}{8}$ inches.

Figure 2.- Continued.



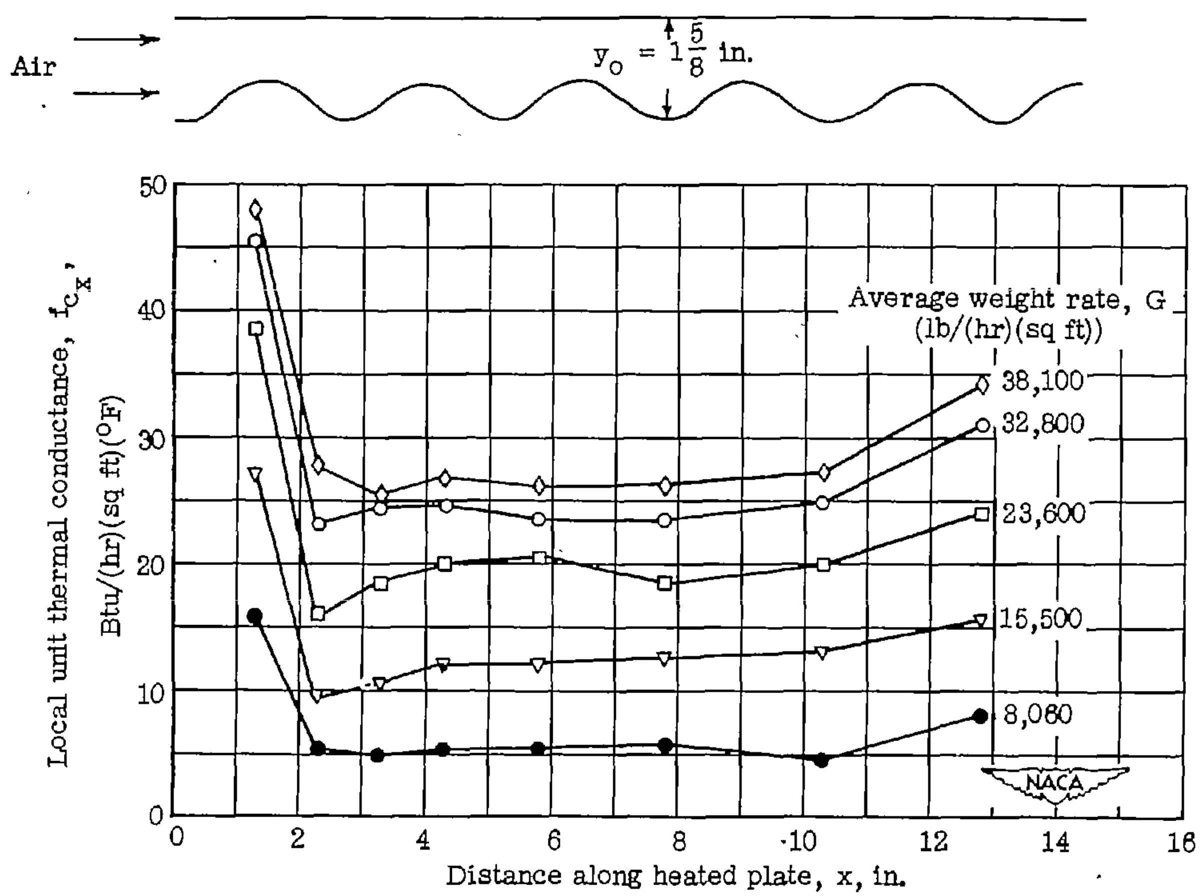
(d) Small wave plate at $y_o = 3\frac{1}{8}$ inches.

Figure 2.- Concluded.



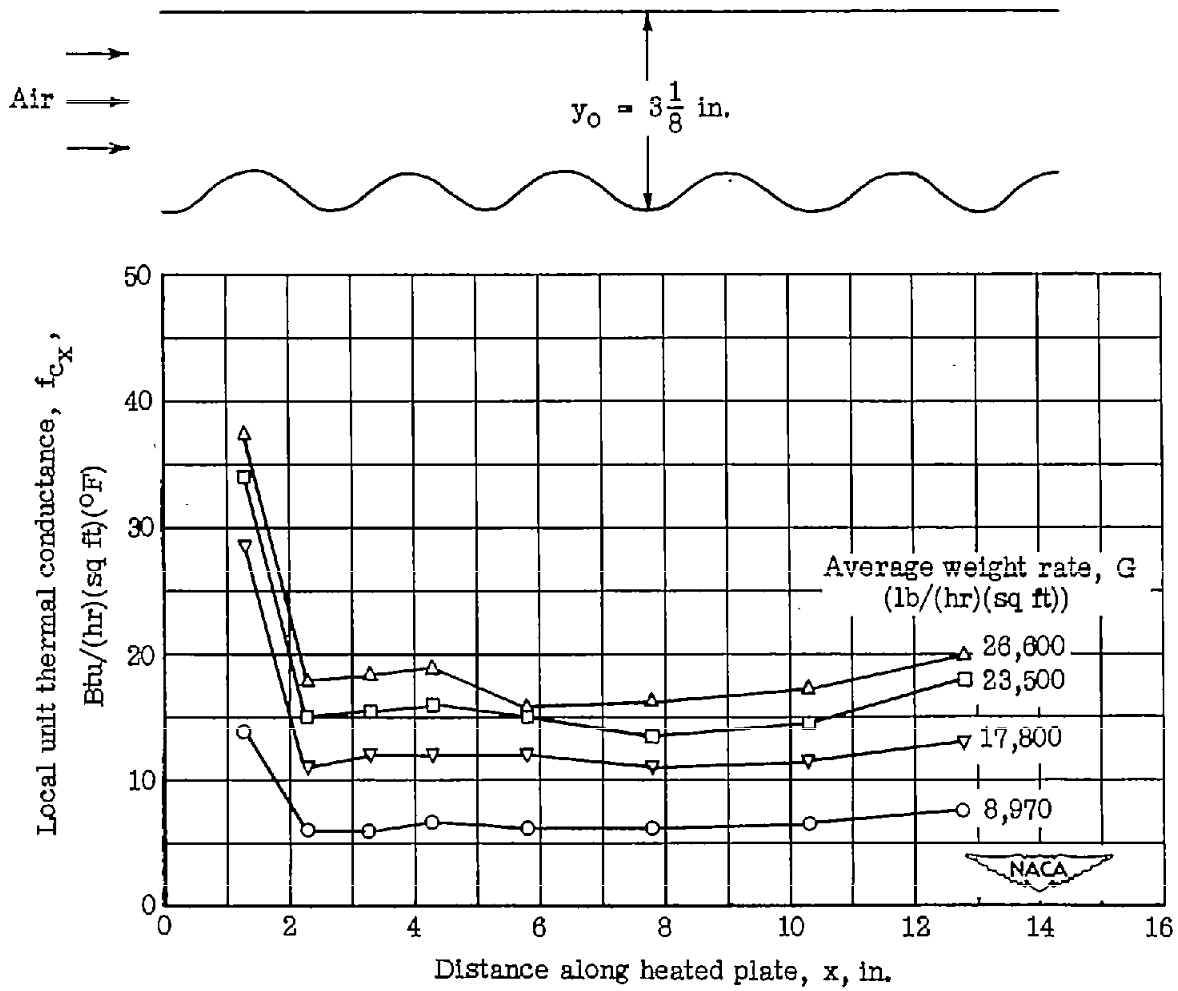
(a) Large wave plate at y_o = 7/8 inch.

Figure 3.- Variation of local unit thermal conductance with distance along heated plate for system (2).



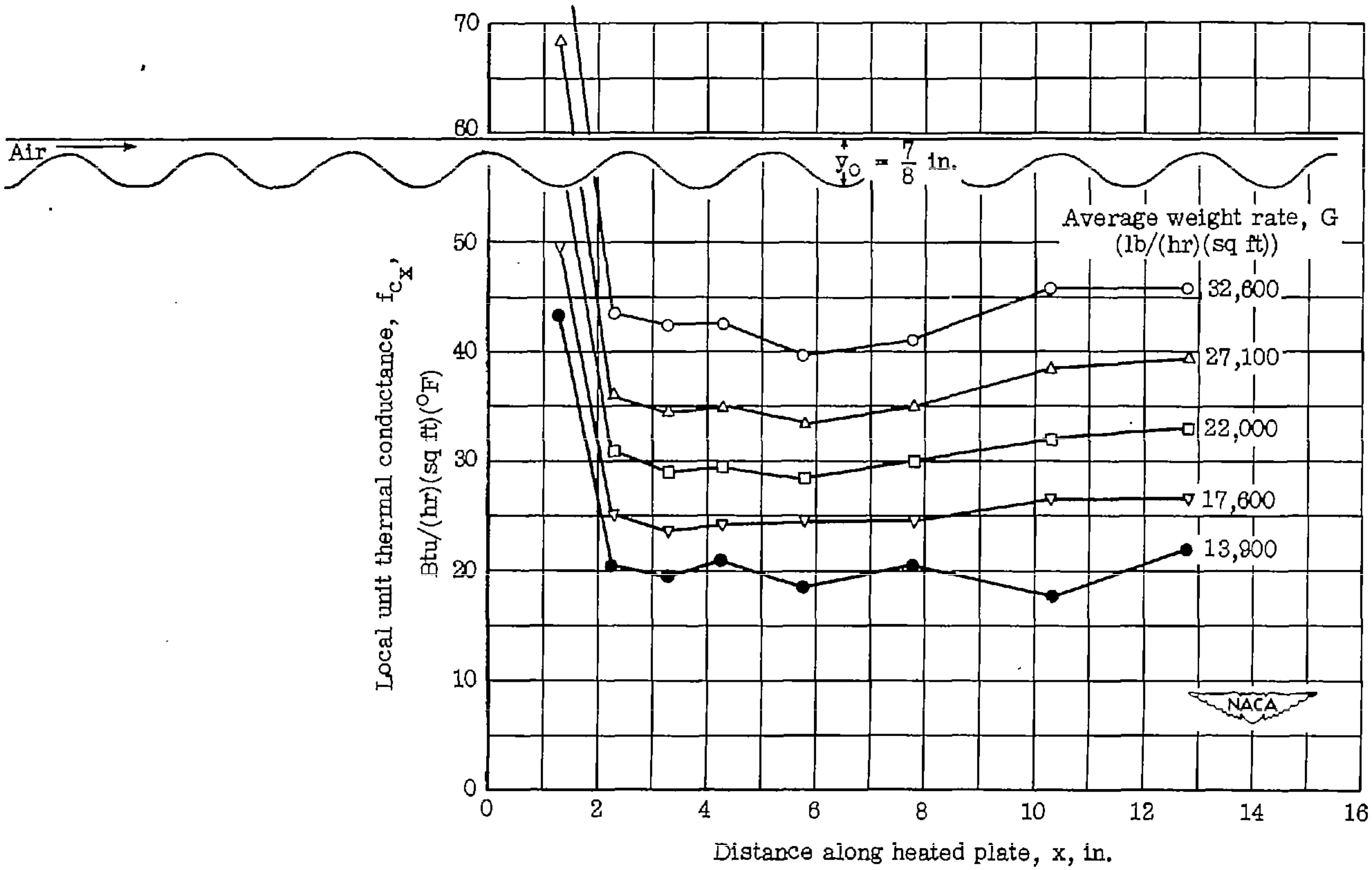
(b) Large wave plate at $y_o = 1\frac{5}{8}$ inches.

Figure 3.- Continued.



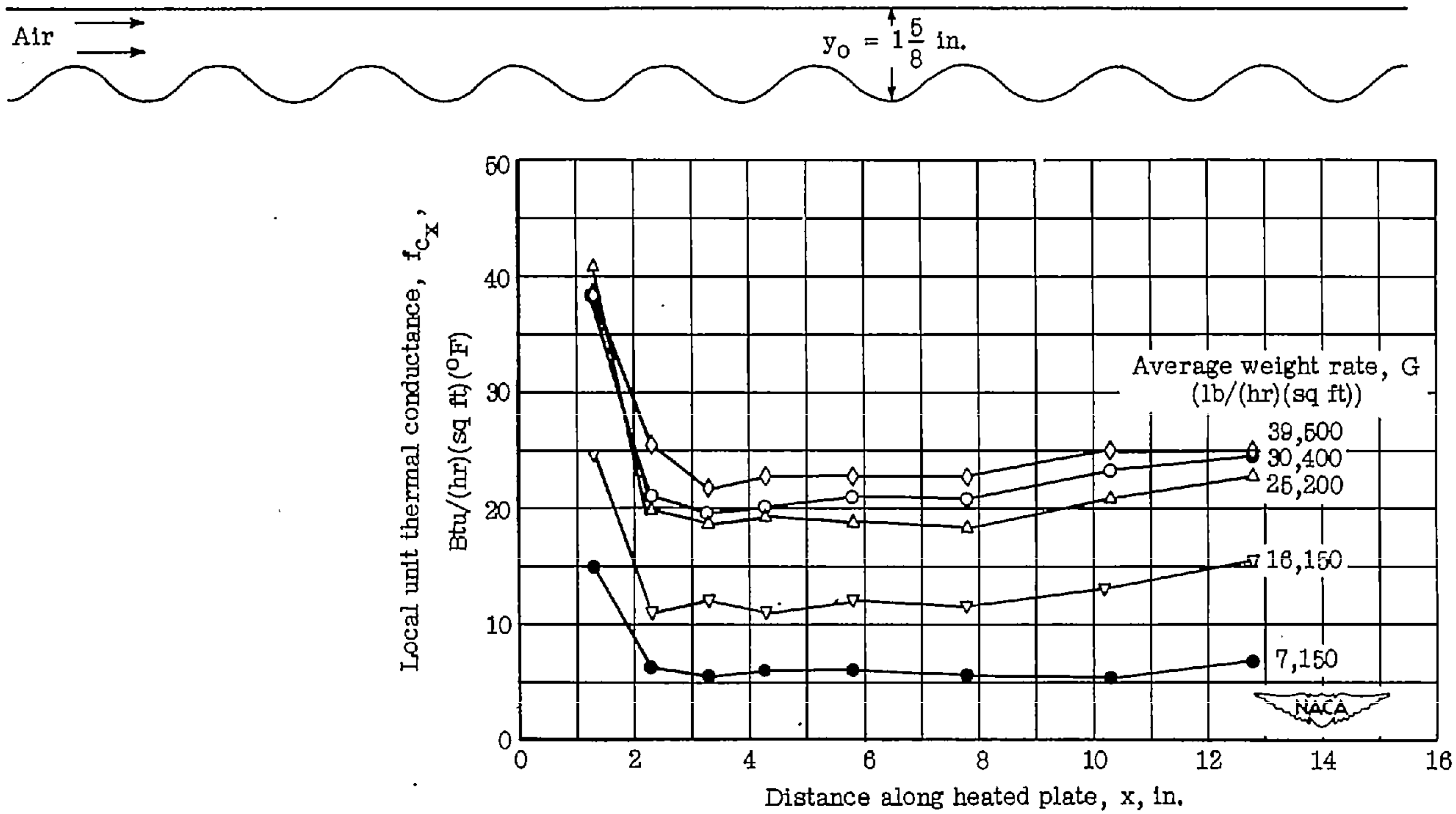
(c) Large wave plate at $y_o = 3\frac{1}{8}$ inches.

Figure 3.- Concluded.



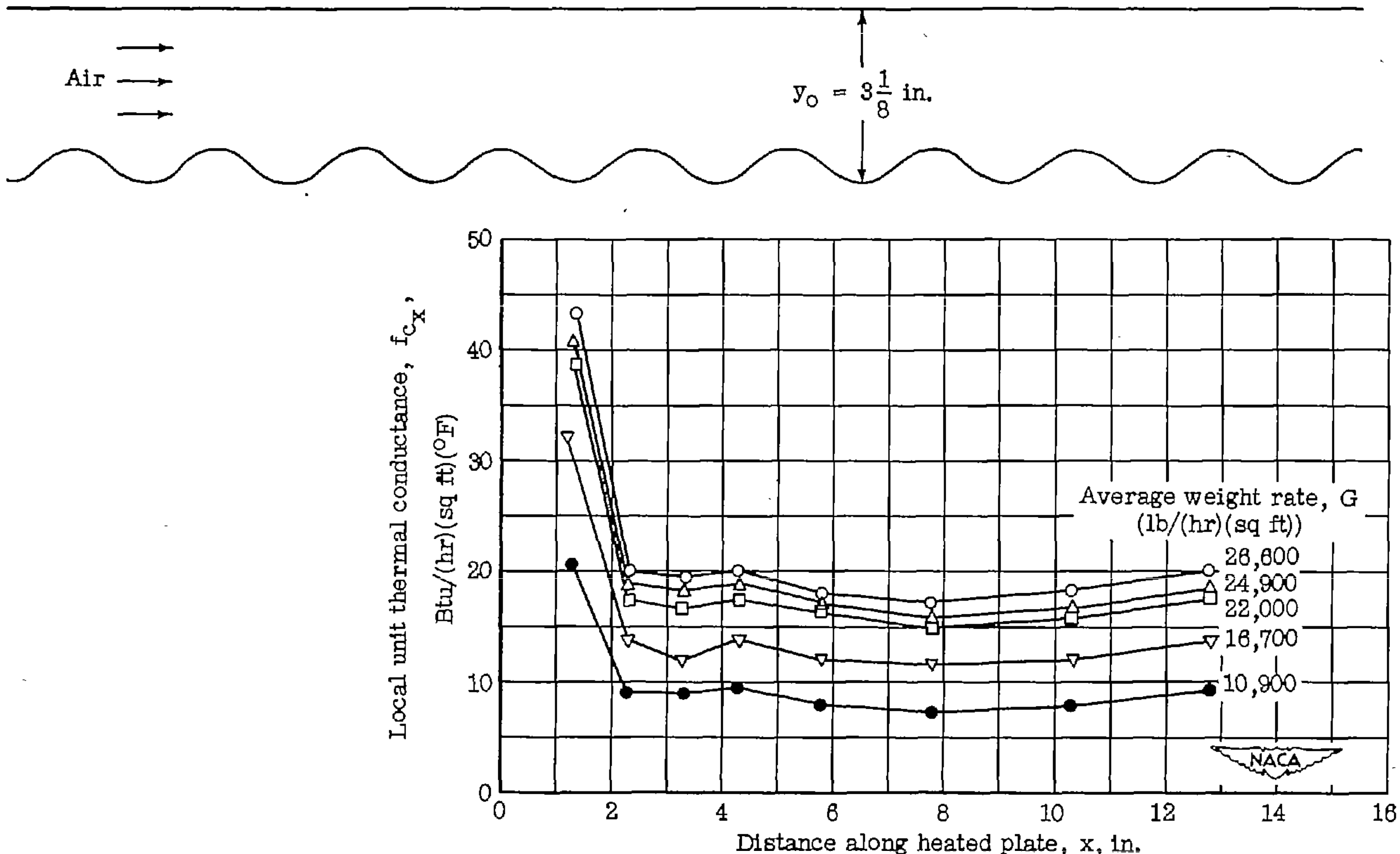
(a) Large wave plate (leading edge upstream) at $y_o = 7/8$ inch.

Figure 4.- Variation of local unit thermal conductance with distance along heated plate for system (3).



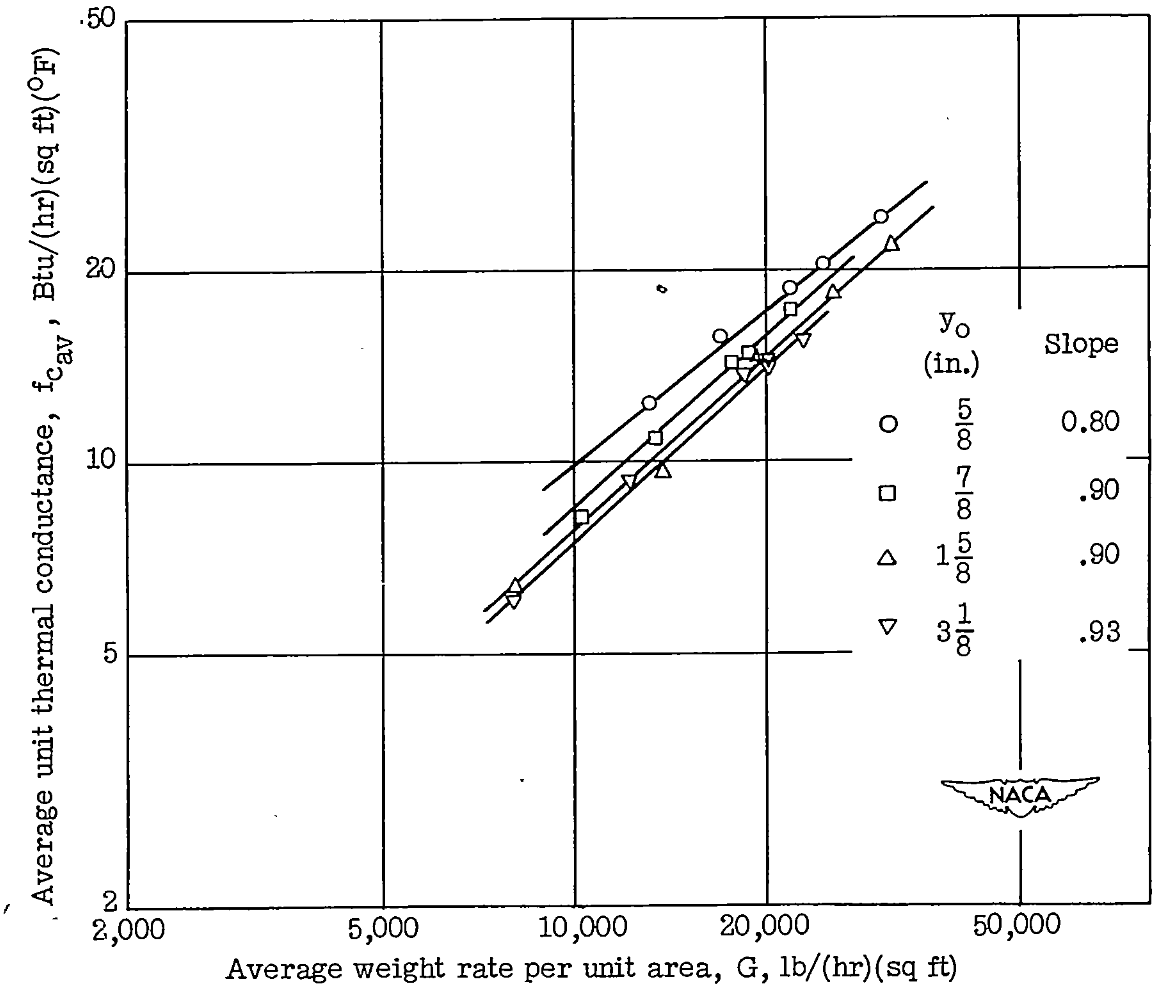
(b) Large wave plate (leading edge upstream) at $y_o = 1\frac{5}{8}$ inches.

Figure 4.- Continued.



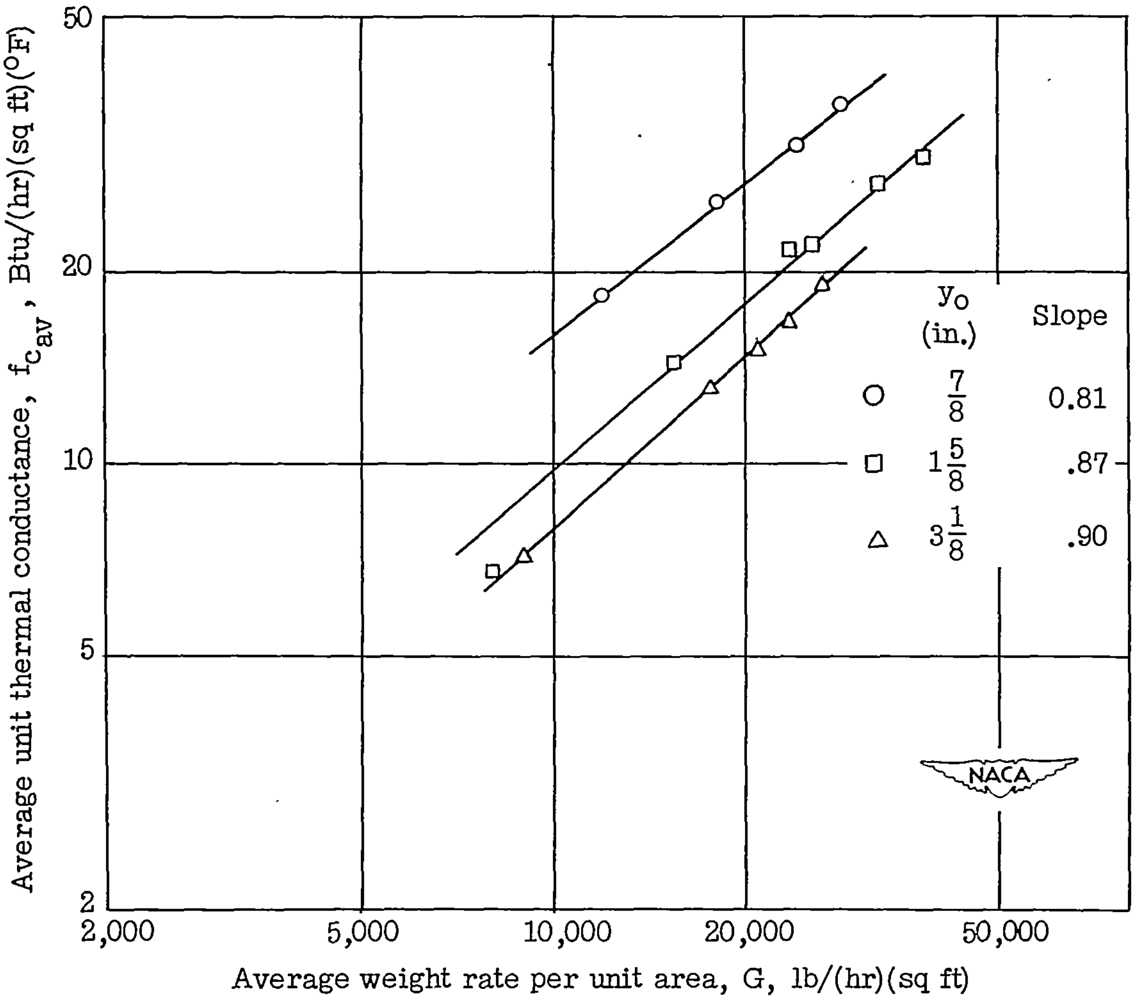
(c) Large wave plate (leading edge upstream) at $y_o = 3\frac{1}{8}$ inches.

Figure 4.- Concluded.



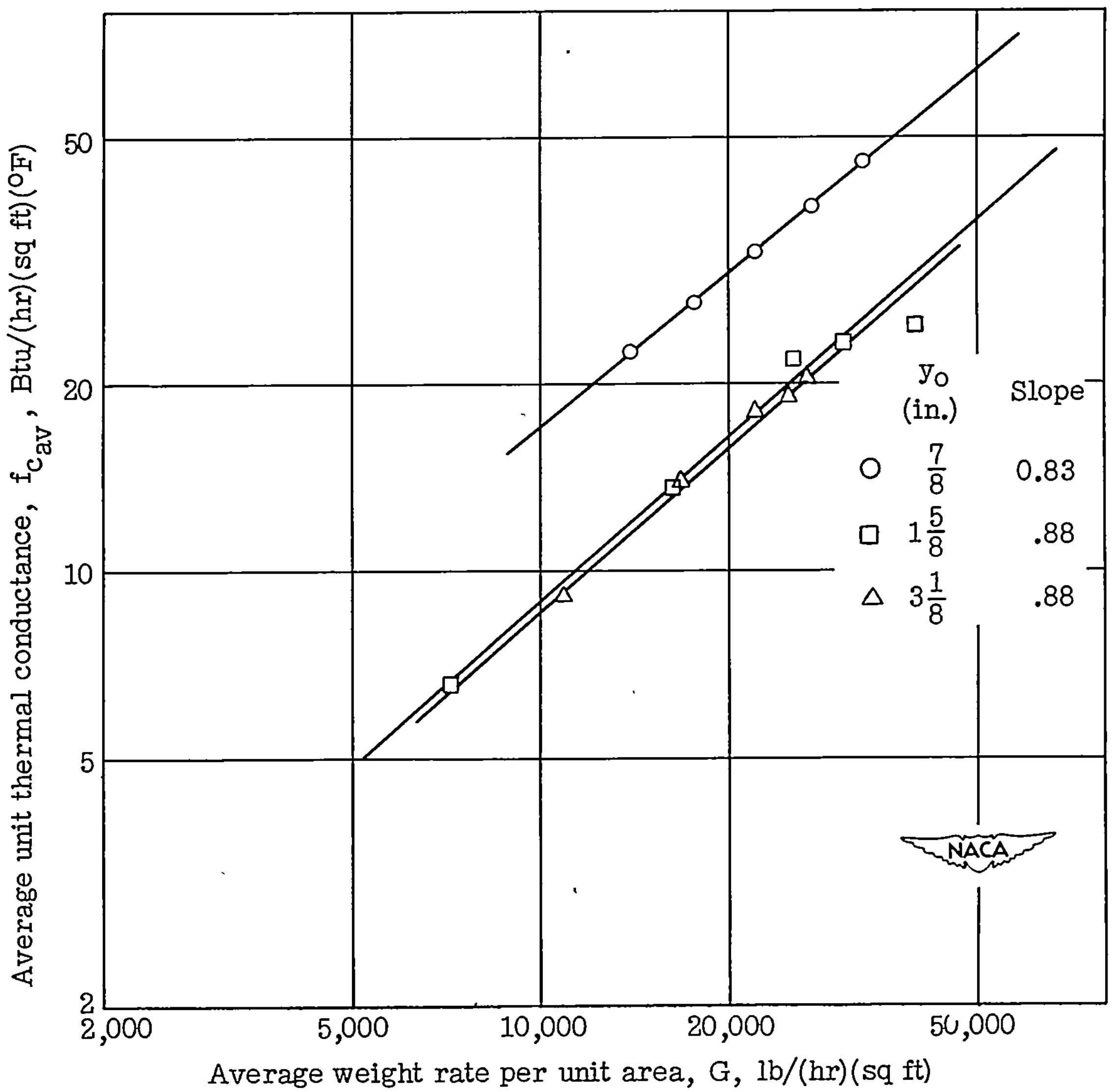
(a) System (1).

Figure 5.- Variation of average unit thermal conductance with average weight rate per unit area.



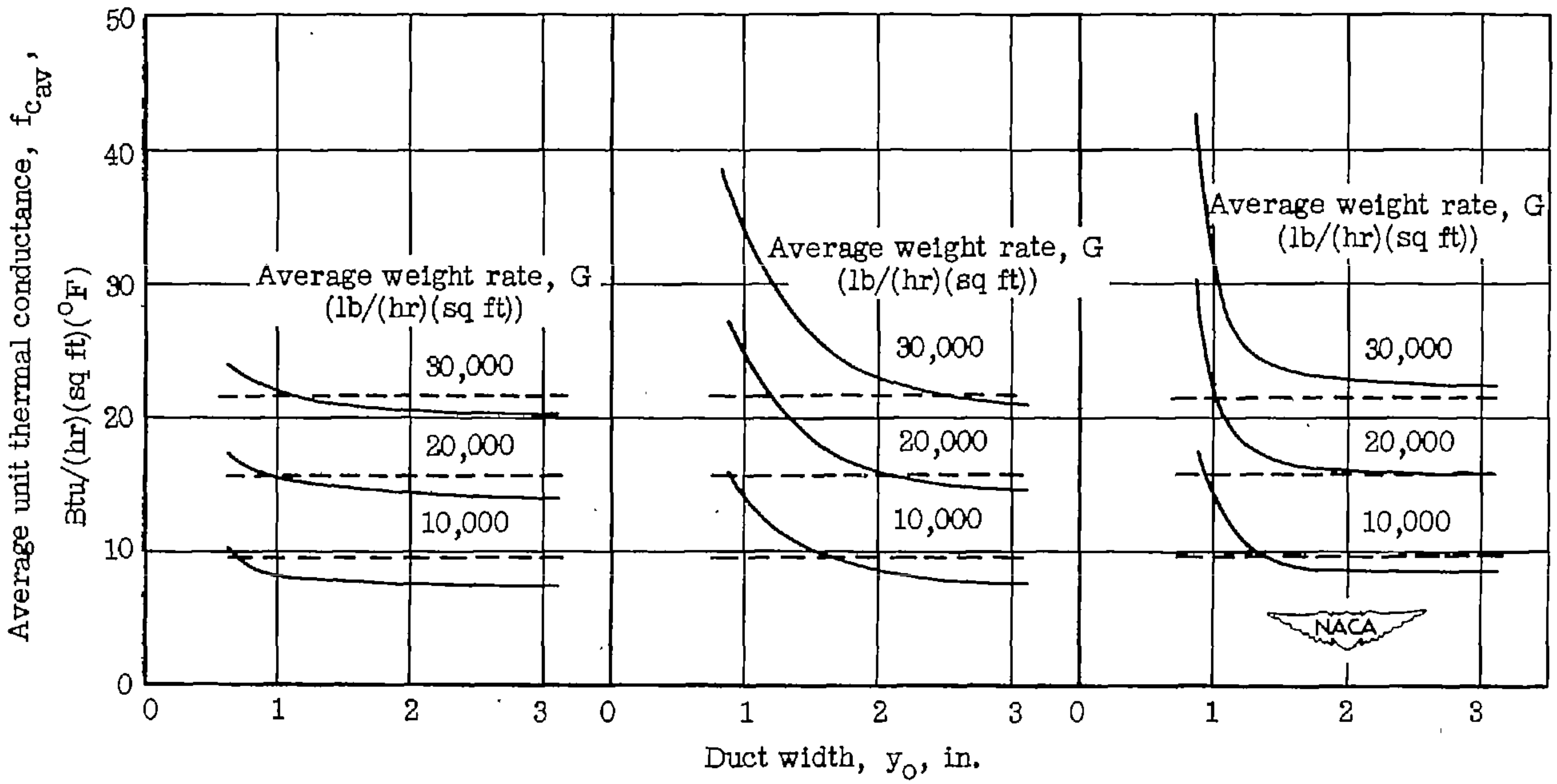
(b) System (2).

Figure 5.- Continued.



(c) System (3).

Figure 5.- Concluded.



(a) System (1). (b) System (2). (c) System (3).

Figure 6.- Variation of average unit thermal conductance with duct width. Dashed curves are values for flat plate (reference 1).

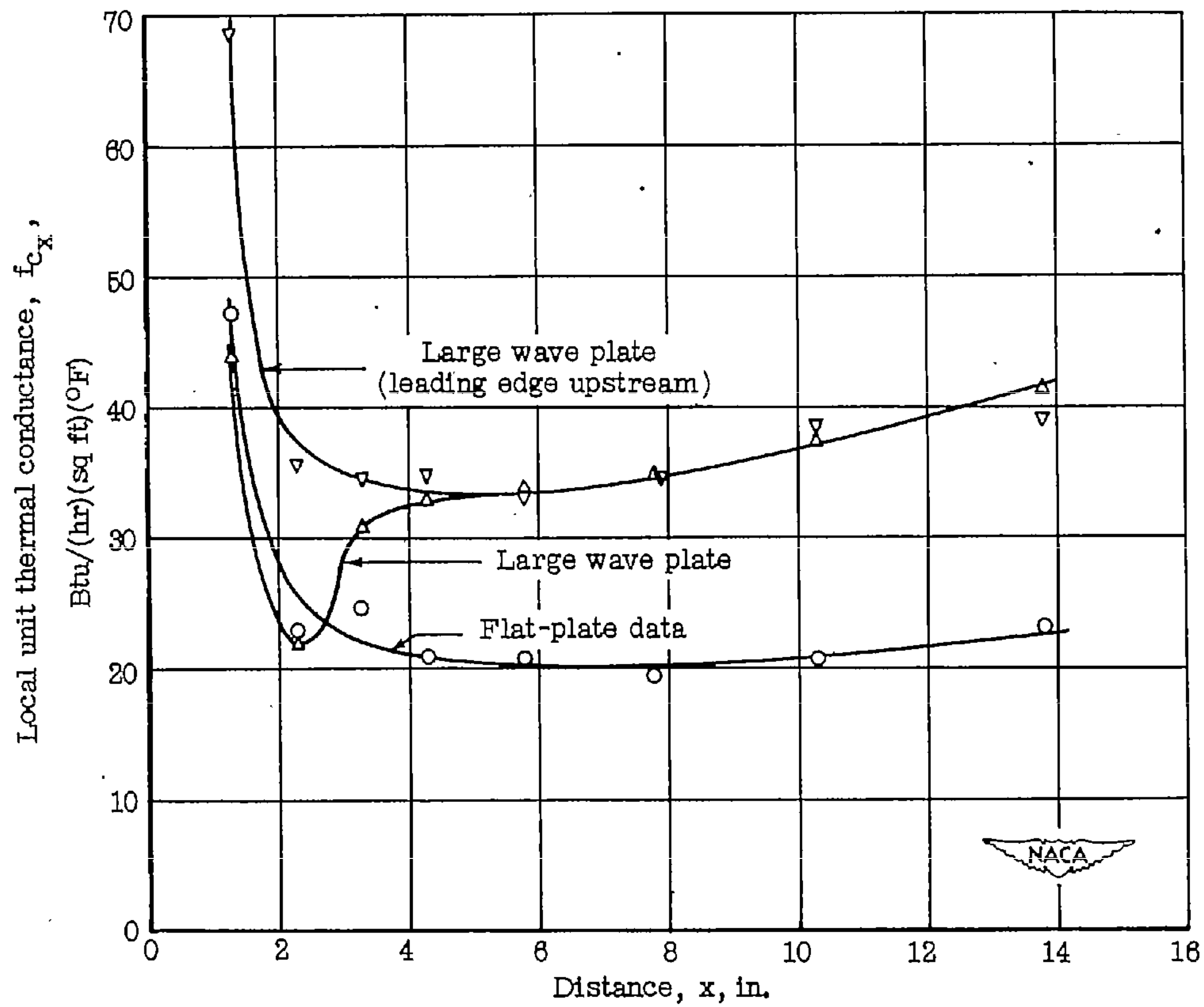
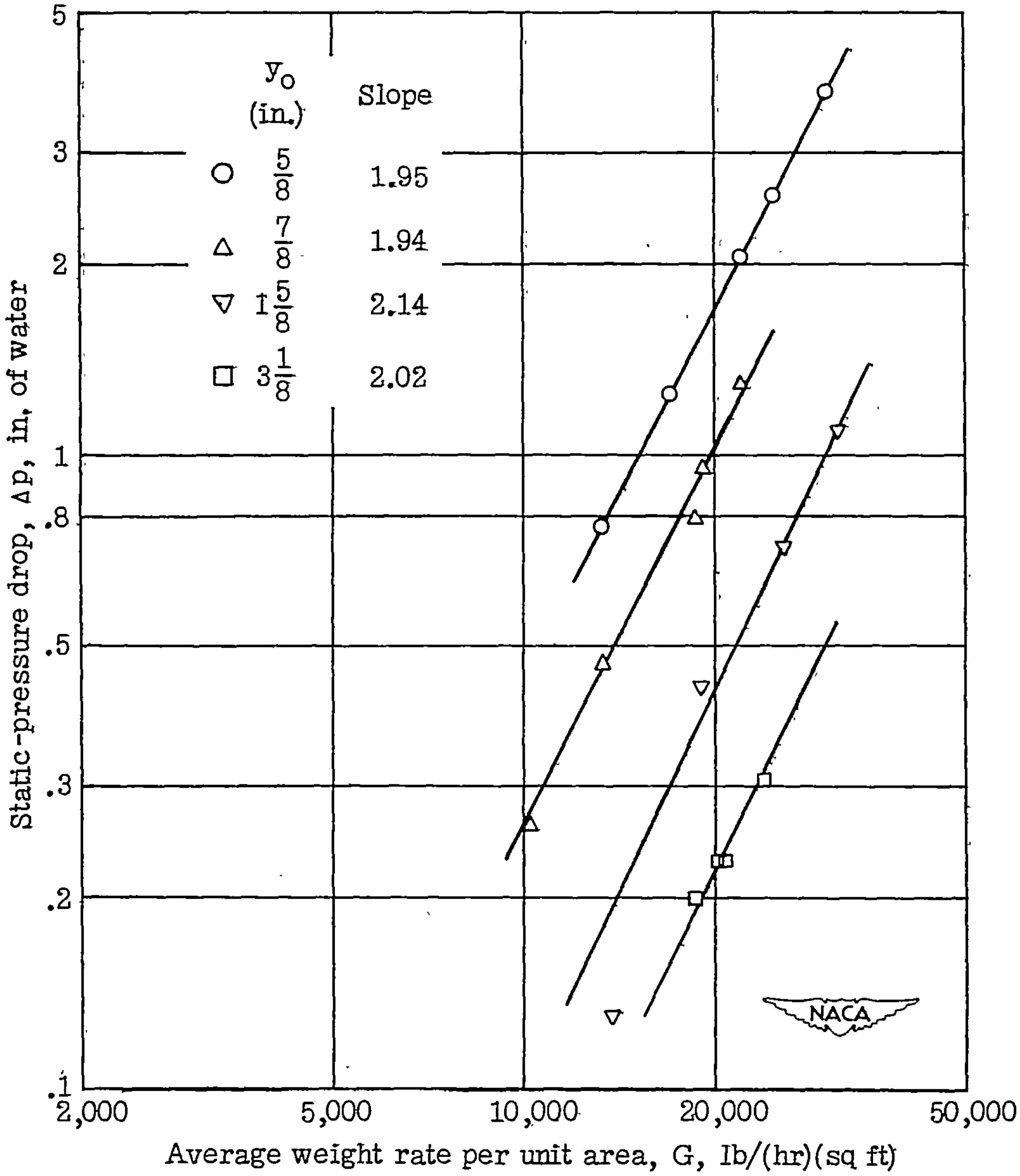
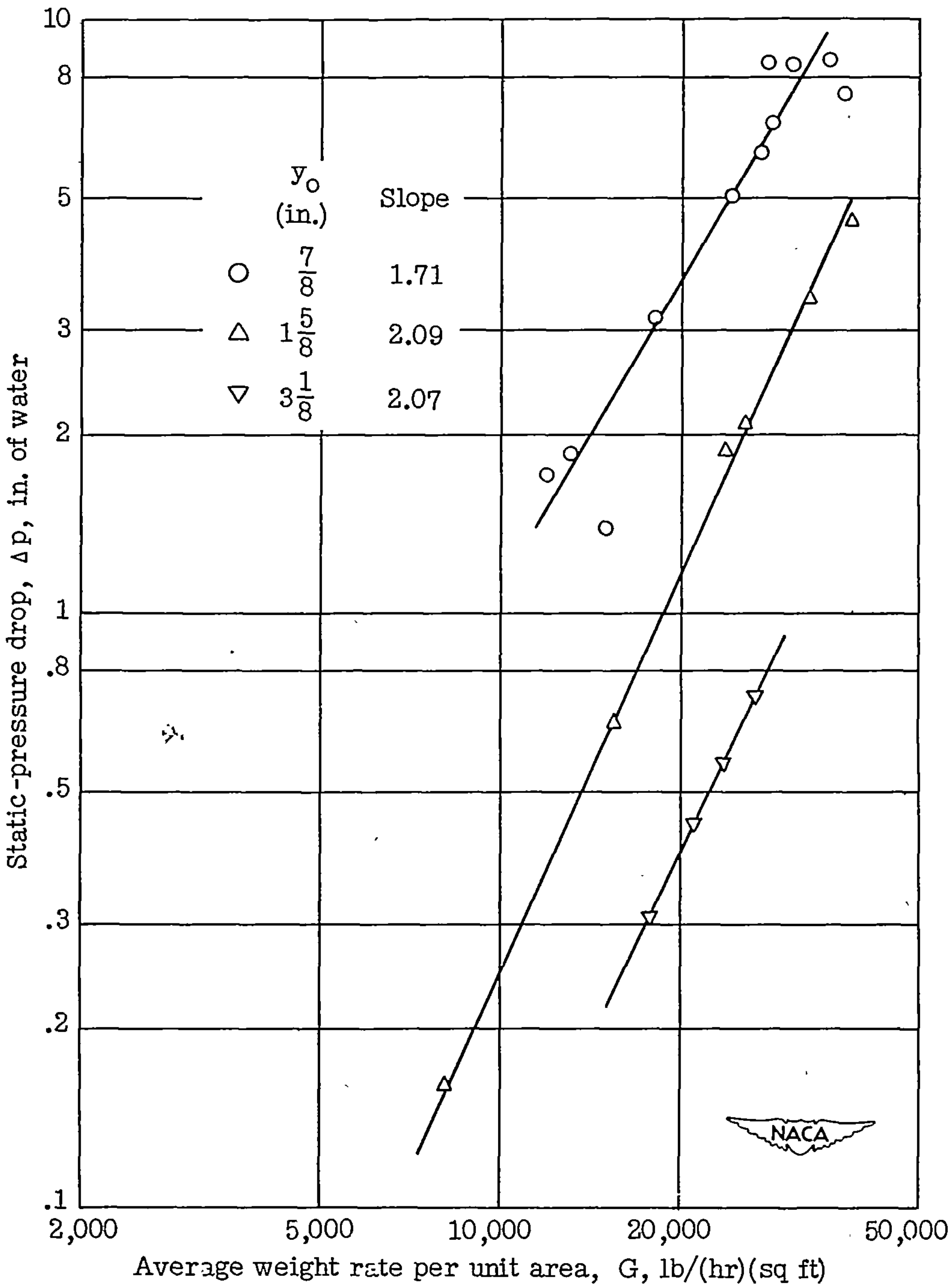


Figure 7.- Comparison of system (2), system (3), and unfinned flat plate for y_0 = 7/8 inch and G ≈ 28,000 pounds per hour per square foot. Flat-plate data from reference 1.



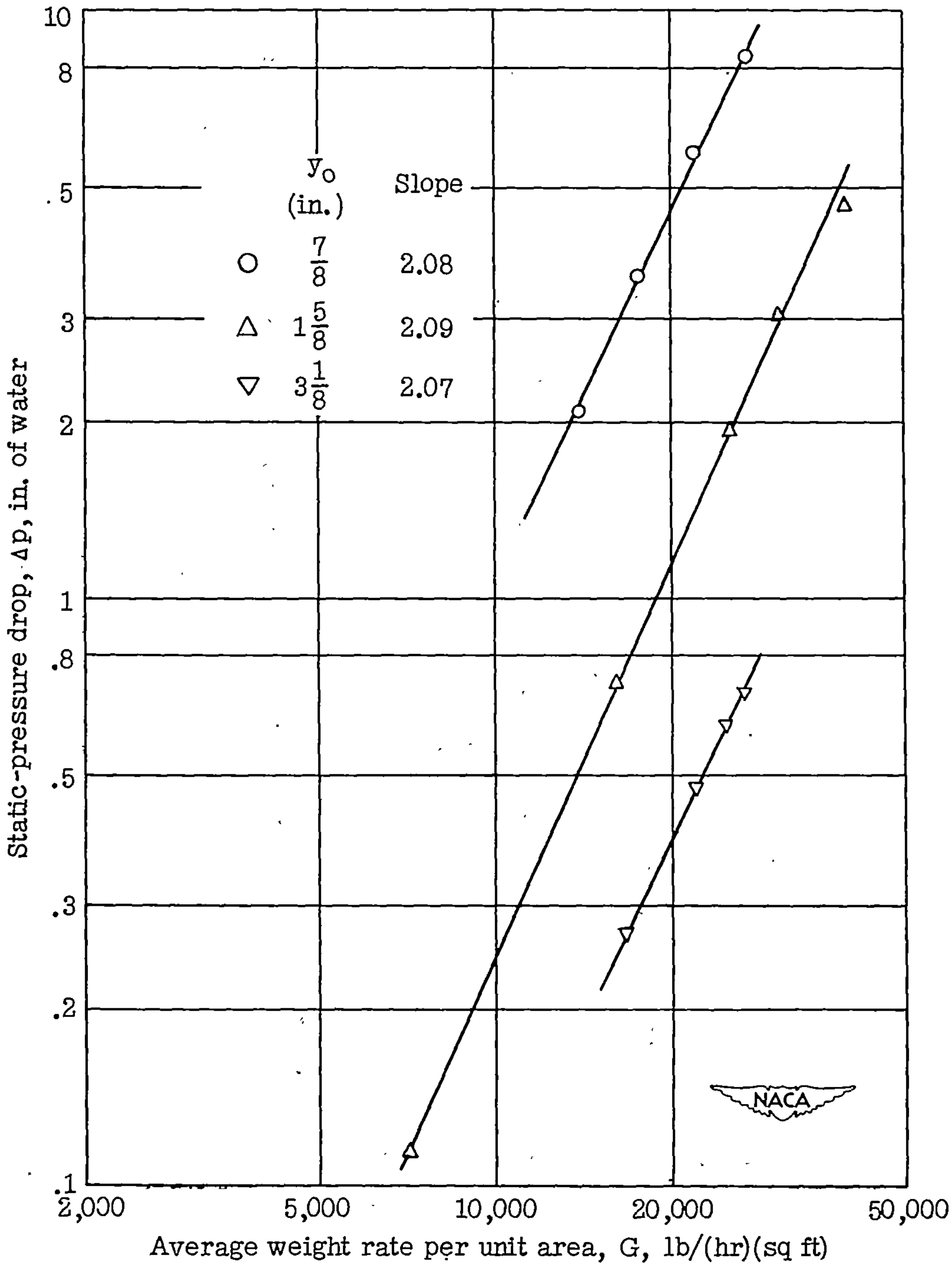
(a) System (1).

Figure 8.- Variation of static-pressure drop with average weight rate per unit area.



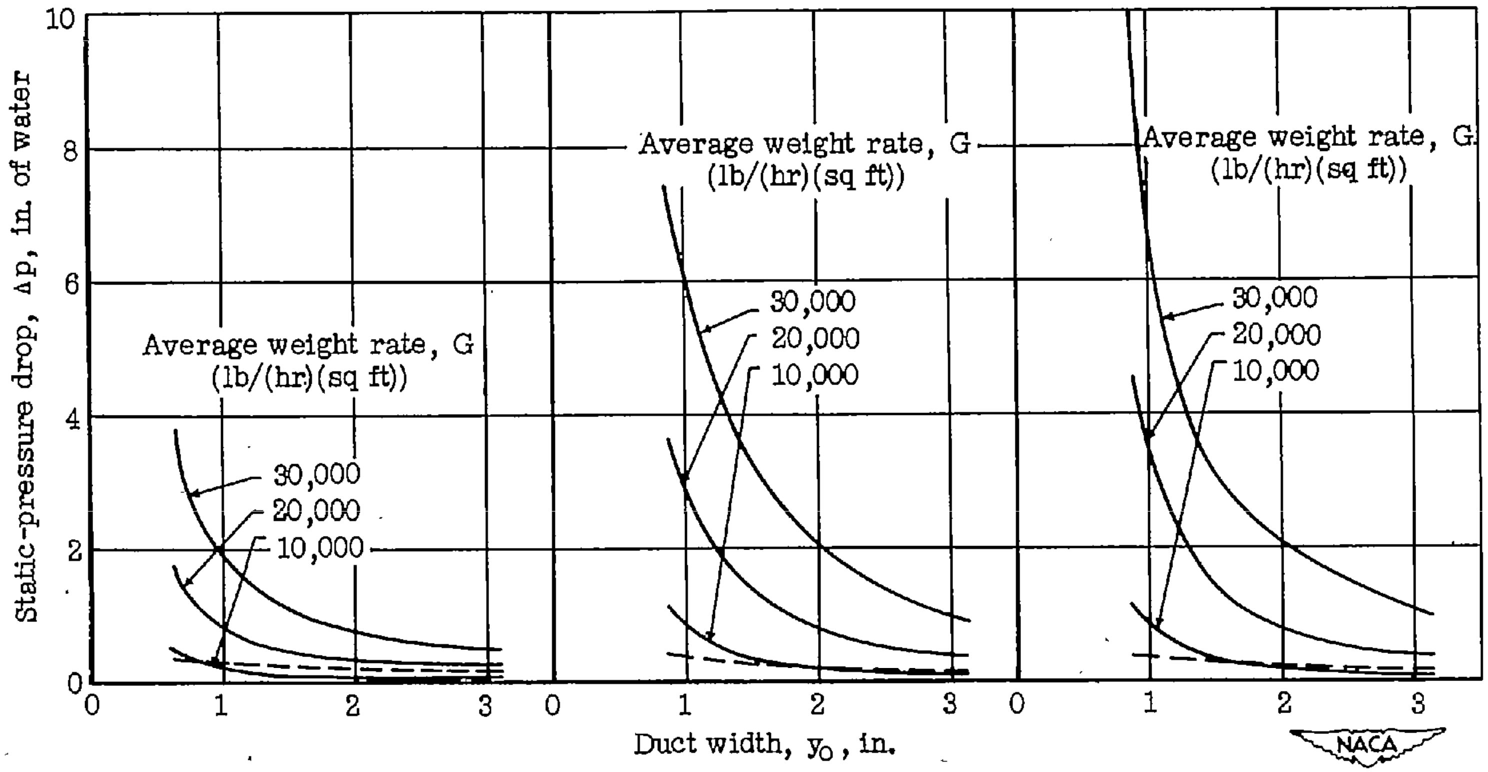
(b) System (2).

Figure 8.- Continued.



(c) System (3).

Figure 8.- Concluded.



(a) System (1). (b) System (2). (c) System (3).

Figure 9.- Variation of static-pressure drop with duct width. Dashed line indicates pressure in flat duct only at average weight rate G of 30,000 pounds per hour per square foot.

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